2

ON THE FEASIBILITY OF A TRANSIENT DYNAMIC DESIGN ANALYSIS METHOD

FINAL TECHNICAL REPORT

May 1, 1991 - August 31, 1993

by

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TABLE OF CONTENTS

EXECUTIVE SUMMARY

This final technical report summarizes the work that was performed on the Navy grant N00014-91-J-4059 since the previous annual report, which was submitted in April, 1992.

The report also includes a summary of the major technical achievments attained during the course of this grant period. More details can be found in the three separate technical reports that were issued during the past two years on those tasks that were identified in the original grant application. There is also a complete listing of the presentations made at professional meetings and papers published as a consequence of the work performed under this grant.

INTRODUCTION

The dynamic behavior of structures subject to mechanical shock loading provides a continuing problem for design engineers concerned with shipboard foundations supporting critical equipment. There are two particular problems associated with shock response that were investigated during the course of the grant period. The first topic explores the possibilities of developing a transient design analysis method that does not degrade the current level of the Navy's shock-proofness requirements for heavy shipboard equipment. The second topic examines the prospects of developing scaling rules for the shock response of simple internal equipment of submarines subject to various attack situations. This second topic was further divided into two tasks: chemical explosive scaling for a given hull; and scaling of equipment response across different hull sizes.

The computer was used as a surrogate shock machine for these studies. The results of taking this approach can provide ideas, suggestions, and scaling rules that could be useful to the Navy. However, it is noted that any numerical results provide only trends in shock design values rather than absolute design values.

STATUS OF PROJECT

The status of the three technical problems which relate to the shock response of shipboard structures investigated under this grant is summarized below.

Feasibility of a Transient Dynamic Design Analysis Method

Since the last reporting period considerable progress was made on this topic. A paper on the initial efforts in developing TDDAM was presented and published in the *Proceedings of the Shock and Vibration Symposium* in October, 1992. This paper was selected and is scheduled to be published in *The Journal of Shock and Vibration*. A new procedure which utilizes an optimization technique has been under study for the purpose of applying the transient dynamic design analysis method (TDDAM) to larger equipment systems than those studied earlier. A technical report on this new approach was issued in May, 1993. Work continued over the summer of 1993 to refine and understand the limitations of this new method. A summary of the results of this recent activity can be found in Appendix A.

Chemical Explosive Scaling for Shock Response of Submarine Equipment

Much of the work on the chemical explosive scaling for shock response of submarine equipment in a given hull was completed in the first year of the grant. Results of the study are found in the technical resport *Scaling for Shock Response of Submarine Equipment* which was issued in April, 1992. During the summer of 1993 a new study was launched on the problem of scaling for two frame-mounted modal oscillators. A summary of the results of this recent activity can be found in Appendix B.

Shock Response of Submarine Equipment Attached to Different Hull Sizes

Work on scaling of the shock response of submarine equipment attached to different hulls was reported in a paper presented and published in the *Proceedings of the Shock and Vibration Symposium* in October, 1992. This paper will also appear in a forthcoming issue of *The Journal of Shock and Vibration*. A technical report was also issued in December, 1992 on the subject. A new paper will be presented at the 64th Shock and Vibration Symposium to be held in Florida in October, 1993 on the most recent developments of the research.

TRANSIENT DYNAMIC DESIGN ANALYSIS METHOD

This investigation sought simple equipment-vehicle models that produce time history responses which are equivalent to the responses that would be achieved using spectral design values employed by the Dynamic Design Analysis Method (DDAM). Two approaches were taken to find the vehicle model to which the equipment model was to be mounted. The direct modelling approach, which was based on the solution of the governing equations of motion, provided solutions to the problem for two-degree of freedom and three-degree of freedom equipment. The vehicle consisted of a single mass which was excited by an impulse and a spring which had one end anchored. The results showed that in the case of the two-degree of freedom equipment, there were multiple solutions. Thus, this result made the important point that there is no unique transient model. It was more difficult to find multiple solutions for the three-degree of freedom equipment models. In most of these cases only one solution was found.

The direct modelling approach becomes impractical for equipment with more than three-degrees of freedom because the algebra becomes unwieldy and there are too many ratios of shock design values to satisfy by trial-and-error. An optimization method, which uses the Simplex Method, was chosen to overcome the shortcomings of the direct modelling method. The optimization method minimizes the error between the time-history responses of modal oscillators and the desired DDAM values. As in the case of the direct modelling approach, the equipment is attached to a vehicle that is excited by an ideal impulse. The vehicle's parameters are optimized so that the absolute maximum (peak) responses of the oscillators closely match the prescribed DDAM inputs. In this approach, the number of degrees of freedom of the vehicle and equipment are not as narrowly limited as in the case of the direct modelling method.

The final phase of this study centered on an examination of the optimization method for possible inherant limitations. Although this method provided acceptable transient models for equipment with as many as nine-degrees of freedom, it was observed in some cases that problems occurred in finding an acceptable transient model for equipment having fewer than nine-degrees of freedom. These problems developed when the frequency content of the equipment's fixed base frequencies reached some upper level. Appendix A is a summary of the results obtained to examine this difficulty. Although limited in scope, these results provide some guidance on the limitations of the method for multiple-degree of freedom equipment.

Two Modal Oscillator Example - The modal weights and the first modal oscillator frequency are fixed as shown in Table A1. Table A2 lists the errors in developing a transient model as the second modal oscillator's frequency, f_2, is increased to as high as 10,000 Hz, where the error is defined as the largest error in a particular mode between the transient pseudo-velocity generated by the optimization method and the corresponding DDAM input value. No significant error is encountered for this example.

Three Modal Oscillator Example - The three modal weights and the first two modal frequencies are shown in Table A3. Varying the third modal frequency, f_3, to as high as 5,000 Hz as shown in Table A4 does not produce a significant error in developing the transient model.

Four Modal Oscillator Example - Tables A5 and A6 show that when the fourth modal frequency, f_4, is increased up to 1,350 Hz, no appreciable error occurs for the transient model. However, when this frequency is raised to 1,500 Hz, an unacceptable error is introduced.

Five Modal Oscillator Example - Table A7 shows the modal characteristics for the five modal oscillator example, where again the highest frequency, f_5, is varied in magnitude as shown in Table A8. Solutions are acceptable up to 300 Hz, and unacceptable for 350 Hz.

Six Modal Oscillator Example - Tables A9 and A10 show acceptable results when the highest modal frequency reaches 150 Hz, and produces unacceptable results at 200 Hz.

Nine Modal Oscillator Example - The same nine modal equipment model reported earlier in the May 1993 Technical Report was reexamined by varying the highest modal frequency f_9. In the earlier study the value of this frequency was fixed at 95 Hz. Table A11 lists the modal characteristics for this example. Table A12 show the results when f_9 is varied. In this case the optimization approach failed when f_9 reached 200 Hz.

It appears from these limited number of examples that as the number of modal oscillators increase beyond three, the success in finding an acceptable transient model by the optimization method is governed by the frequency content.

CHEMICAL EXPLOSIVE SCALING FOR SHOCK RESPONSE OF SUBMARINE EQUIPMENT

The results of this study demonstrated that useful information may be obtained by using a computer as an initial surrogate for shock testing purposes. These results have shown the relative changes in shock design values for different boats and attack geometries. It is emphasized that the test sections were small in size and devoid of typical equipment present in a real compartment. Consequently, the results provide only trends in shock design values rather than absolute design numbers.

Large amounts of computer generated data were collected for two submarine models each of which contained a single-degree of freedom equipment. The study attempted to develop some scaling rules for handeling field data that may exist for a given class of boat. The intent of these scaling rules is to allow greater useage of these data for different equipment subject to a variety of charge weights. A scaling rule was developed that includes the charge weight, the equipment weight, and the equipment frequency. The general scaling rule was applied in two different ways: a linear least squares fit through the data; and a parabolic least squares fit through the data. The latter approach provided better results in predicting equipment response over a wide range of the shock factors.

During the final phase of the grant period a two-degree of freedom system represented by two modal oscillators was studied to determine if the scaling rules obtained for a single oscillator were applicable. In particular, the following scaling rule was examined:

$$s_b = s_a \left(\frac{Q_a}{Q_b} \right)^n \tag{1}$$

in which

s_a = response slope associated with charge weight Q_a

s_b = predicted response slope for charge weight Q_b

n = 0.125 for the single oscillator for $600 \le Q \le 3625$ lb TNT

The question is whether the exponent n in eq.(1) changes due to the presence of the second oscillator. A summary of these new results for two oscillators are found in the tables of data in Appendix B. For example, consider Table B1(a) that was developed for one oscillator weighing 35 kips at 15 Hz frequency, while the second oscillator's weight varied from 5 to 30 kips and the frequency varied from 20 to 35 Hz as shown in the table. The data in the table list the values of the exponent n obtained from the actual response data that would be used in eq.(1) to predict the response of oscillator 1. It appears that as the second oscillator weight and frequency increase in magnitude, the values of n decrease toward negative values. The dashed lines in Table B1(a) is introduced to separate the data into two regions; the upper left-hand portion that produces acceptable values of n for using the above scaling rule, and the lower right portion that contains values of n that are not acceptable. The average value of n in the acceptable region of the table is n = 0.0917. Substituting this value for n into eq.(1) for the case where the largest and lowest charge weights are used, i.e., 3625 lb and 600 lb of TNT, in place of the lowest value of n = 0.07207 shown in the acceptable region of Table B1(a) produces the largest error of 3.6%. Table B1(b) shows the distribution of the n values for predicting the response for the second oscillator. Interestingly, the data show that the range in oscillator 2's modal weight and frequency produce acceptable values of n throughout the weight and frequency range shown in Table B2(b) with an average value of n = 0.0930.

Tables B2(a) shows similar results for the case where oscillator 1 was 25 kips, 15 Hz. In this case, the recommended region, as marked off in the table, for using eq.(1) produces an average value of n = 0.0905 for oscillator 1. This produces an error of 3.99% in eq.(1) for the largest charge weight ratio and when n = 0.06876. An average value of n = 0.0967 would provide acceptable predictions using eq.(1) for the range in the modal weights and frequencies for oscillator 2 as shown in Table B2(b).

Table B3(a) lists the results for oscillator 1 being 15 kips, 15 Hz and oscillator 2's modal weight being 5 and 10 kips, respectively. The acceptable region is marked off as in the other examples, where the average value of n = 0.1037 would be acceptable. As in the other cases, an average value of n = 0.1048 would suffice for the range in oscillator 2 as shown in Fig. B3(b).

SCALING FOR SHOCK RESPONSE OF SUBMARINE EQUIPMENT ATTACHED TO DIFFERENT HULL SIZES

This study developed a scaling procedure for the case where response data from a given sized hull could be scaled to predict the response in the variation of the charge weight, equipment weight, and/or the equipment frequency that might occur in a different, but linearly scaled, hull. This procedure is carried out in two steps: in the first step the hull and the equipment in the original hull are linearly scaled to the desired new hull. The conventional linear scaling law predicts the response using the response data for the original hull. The second step utilzed the general scaling rule for variations in the charge weight, equipment weight, and/of the equipment frequency. The results in predicting the responses in this way were well within acceptable engineering accuracy.

SUMMARY OF PUBLICATIONS

1. O'Hara, G.J., and Cunniff, P.F., "Time History Analysis of Systems as an Alternative to a DDAM-Type Analysis," *Proceedings of the 62nd Shock & Vibration Symposium*, Oct.,1991.

2. O'Hara, G.J., and Cunniff, P.F., "Scaling for Shock Response of Submarine Equipment," University of Maryland Technical Report, April, 1992.

3. Cunniff, P.F., "Some Research Topics at the University of Maryland on the Design of Equipment in a Shock Environment," presented at the <u>ONR Workshop on Underwater Explosion Effects on Structures and Shock Mitigation</u>, University of Maryland, College Park, Sept.,1992.

4. Cunniff, P.F., and O'Hara, G.J., "On the Feasibility of a Transient Dynamic Design Analysis Method," *Proceedings of the 63rd Shock & Vibration Symposium*, Oct., 1992. To be published in *The Journal of Shock & Vibration*.

5. O'Hara, G.J., and Cunniff, P.F., "Scaling for Shock Response of Equipment in Different Submarines," *Proceedings of the 63rd Shock & Vibration Symposium*, Oct., 1992. To be published in *The Journal of Shock & Vibration*.

6. Cunniff, P.F., and O'Hara, G.J., "Scaling for Shock Response of Equipment in Different Submarines," University of Maryland Technical Report, Dec., 1992.

7. Cunniff, P.F., and O'Hara, G.J., "Some Thoughts on Perceived DDAM Problems," *Proceedings of the 11th International Modal Analysis Conference*, Feb., 1993.

8. Cunniff, P.F., and Pohland, R.D., "On the Feasibility of a Transient Dynamic Design Analysis," University of Maryland Technical Resport, May, 1993.

9. Cunniff, P.F., O'Hara, G.J., and Wagner, R.J., "Parabolic Scaling for Shock Response of Equipment in Different Submarines," to be presented at the 64th Shock & Vibration Symposium,)ct. 1993, and to be published in the *Proceedings*.

ACKNOWLEDGMENTS

Robert Pohland and Robert Wagner, mechanical engineering graduate students, are recognized for their assistance with the work carried out under this grant.

APPENDIX A - Recent Developments on the Dynamic Design Analysis Method

To examine the limitations of the transient DDAM, examples of equipment with as many as nine degrees of freedom attached to given vehicles were solved using the optimization codes. For each specified number of equipment DOF, the largest modal frequency was increased systematically in an effort to determine whether the transient DDAM procedure has inherent limitations with regard to large modal frequencies.

<u>Two Modal Oscillator Example:</u>

Table A1 - Two DOF Example

i	1	2
Modal Weight (kips)	100	50
Frequency (Hz)	75	f_2

Table A2 lists the largest errors encountered in the transient models as the second modal frequency was increased. Vehicle 2 was used in the optimization (refer to Fig.A1).

Table A2 - Errors Associated with the Two DOF Example

f_2 (Hz)	Largest Modal Oscillator Error (%)
500	0.00
1000	0.00
2500	0.00
5000	0.01
10000	0.02

<u>Three Modal Oscillator Example:</u>

Table A3 - Three DOF Example

i	1	2	3
Modal Weight (kips)	100	50	10
Frequency (Hz)	25	75	f_3

Table A4 lists the largest errors encountered in the transient models as the third modal frequency was increased. Vehicle 4 was used in the optimization (refer to Fig.A2).

Table A4 - Errors Associated with the Three DOF Example

f_3 (Hz)	Largest Modal Oscillator Error (%)
500	0.04
1000	0.01
2500	0.01
5000	0.00

Four Modal Oscillator Example:

Table A5 - Four DOF Example

i	1	2	3	4
Modal Weight (kips)	100	70	30	10
Frequency (Hz)	25	50	75	f_4

Table A6 lists the largest errors encountered in the transient models as the fourth modal frequency was increased. Vehicle 6 was used in the optimization (refer to Fig.A3).

Table A6 - Errors Associated with the Four DOF Example

f_4 (Hz)	Largest Modal Oscillator Error (%)
500	0.02
1000	0.50
1350	0.06
1500	74.57

Five Modal Oscillator Example:

Table A7 - Five DOF Example

i	1	2	3	4	5
Modal Weight (kips)	100	80	60	30	10
Frequency (Hz)	10	25	50	75	f_5

Table A8 lists the largest errors encountered in the transient models as the fifth modal frequency was increased. Vehicle 6 was used in the optimization.

Table A8 - Errors Associated with the Five DOF Example

f_5 (Hz)	Largest Modal Oscillator Error (%)
250	0.03
300	0.51
350	23.78

Six Modal Oscillator Example:

Table A9 - Six DOF Example

i	1	2	3	4	5	6
Modal Weight (kips)	100	80	60	50	30	10
Frequency (Hz)	10	15	25	50	75	f_6

Table A10 lists the largest errors encountered in the transient models as the sixth modal frequency was increased. Vehicle 6 was used in the optimization.

Table A10 - Errors Associated with the Six DOF Example

f_6 (Hz)	Largest Modal Oscillator Error (%)
100	1.70
150	3.08
200	14.58

Nine Modal Oscillator Example:

Table A11 - Nine DOF Example

i	1	2	3	4	5	6	7	8	9
Modal Weight (kips)	100	90	75	60	45	26	15	10	6
Frequency (Hz)	20	25	40	58	61	70	75	90	f_9

Table A12 lists the largest errors encountered in the transient models as the ninth modal frequency was increased. Vehicle 6 was used in the optimization.

Table A12 - Errors Associated with the Nine DOF Example

f_9 (Hz)	Largest Modal Oscillator Error (%)
100	3.77
125	4.27
150	4.55
175	3.00
200	25.14

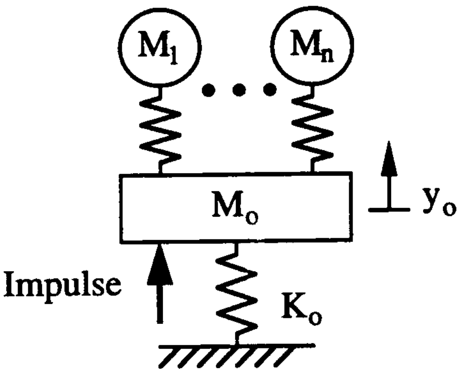

Figure A1 - Vehicle 2

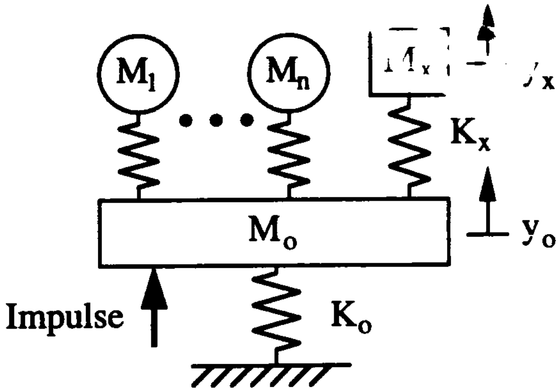

Figure A2 - Vehicle 4

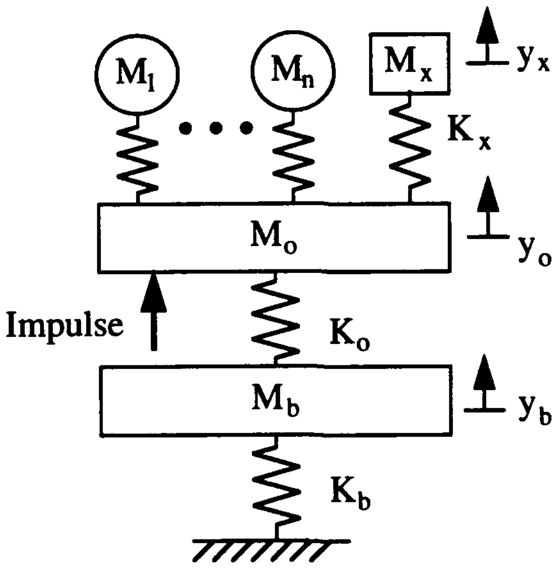

Figure A3 - Vehicle 6

APPENDIX B - Recent Developments on Chemical Explosive Scaling for Shock Response of Submarine Structures

To examine the applicability of the scaling rule used in eq. (1) for two-degree of freedom systems, the value of the exponent n was determined for several two-degree of freedom examples. The values of n shown in the following tables are the average values for n found with Q_a = 1160 lb charge in eq. (1).

For each example, the first table shows the distribution of the values of the exponent n for oscillator 1 with varying weights and frequencies for oscillator 2. The second table shows the distribution of the values of the exponent n for the varying weights and frequencies of oscillator 2 that are used in the example.

Example with 35 kip, 15 Hz Oscillator 1:

Table B1(a) - n Distribution of Oscillator 1
Oscillator 1: 35 kips, 15 Hz

Osc 2	20 Hz	25 Hz	30 Hz	35 Hz
5 kips	0.10783	0.09984	0.09494	0.09743
10 kips	0.09927	0.08996	0.07877	0.07824
15 kips	0.09765	0.08179	0.05918	0.02128
20 kips	0.09666	0.07207	-0.02657	-0.12750
25 kips	0.09540	0.04398	-0.12350	-0.17320
30 kips	0.09390	-0.01394	-0.16668	-0.15784

Table B1(b) - n Distribution of Oscillator 2
Oscillator 1: 35 kips, 15 Hz

Osc 2	20 Hz	25 Hz	30 Hz	35 Hz
5 kips	0.10520	0.09663	0.09688	0.09641
10 kips	0.09634	0.09686	0.09495	0.09191
15 kips	0.09707	0.09519	0.09164	0.09020
20 kips	0.09655	0.09305	0.09061	0.08722
25 kips	0.09507	0.09075	0.08892	0.08545
30 kips	0.09388	0.09055	0.08677	0.08429

Example with 25 kip, 15 Hz Oscillator 1:

Table B2(a) - n Distribution of Oscillator 1
Oscillator 1: 25 kips, 15 Hz

Osc 2	20 Hz	25 Hz	30 Hz	35 Hz
5 kips	0.11066	0.10241	0.09755	0.09734
10 kips	0.10486	0.08158	0.06902	0.06876
15 kips	0.09760	0.07098	0.02061	-0.02404
20 kips	0.09442	0.05901	-0.10044	-0.16000

Table B2(b) - n Distribution of Oscillator 2
Oscillator 1: 25 kips, 15 Hz

Osc 2	20 Hz	25 Hz	30 Hz	35 Hz
5 kips	0.11116	0.10532	0.09630	0.09444
10 kips	0.10637	0.09401	0.09659	0.09380
15 kips	0.09748	0.09668	0.09350	0.09058
20 kips	0.09693	0.09452	0.09068	0.08870

Example with 15 kip, 15 Hz Oscillator 1:

Table B3(a) - n Distribution of Oscillator 1
Oscillator 1: 15 kips, 15 Hz

Osc 2	20 Hz	25 Hz	30 Hz	35 Hz
5 kips	0.11768	0.10954	0.10020	0.10284
10 kips	0.10692	0.08485	0.06551	0.06376

Table B3(b) - n Distribution of Oscillator 2
Oscillator 1: 15 kips, 15 Hz

Osc 2	20 Hz	25 Hz	30 Hz	35 Hz
5 kips	0.11670	0.11472	0.10482	0.09477
10 kips	0.11218	0.10303	0.09691	0.09528